CHAPTER 1
Classification

Consistent core geometry in heat exchangers

1.1 Class definition

Direct-sizing is concerned with members of the class of heat exchangers that have consistent geometry throughout the exchanger core, such that local geometry is fully representative of the whole surface. The following configurations are included in that class and are discussed further in this chapter, but the list is short and illustrative only, namely:

- Helical-tube, multi-start coil
- Plate–fin
- RODbaffle
- Helically twisted, flattened tube
- Spirally wire-wrapped
- Bayonet tube
- Wire-woven tubes
- Porous matrix heat exchanger

Illustrations of many types of exchanger are included in the following recent texts:

- Hewitt *et al.* (1994), Chapter 4
- Hesselgreaves (2001), Chapter 2
- Shah & Sekulic (2003), Chapter 1

1.2 Exclusions and extensions

Exclusions

Not every heat exchanger design is considered in this textbook, for the main objective is to study thermal design of contraflow exchangers proceeding via steady-state direct-sizing, through optimization, to the study of transients.

Most automotive heat exchangers operate in crossflow, and have a relatively small flow length on the air-side. They may be constructed of tubes inserted in corrugated plate–fins, or made up from welded channels with corrugated fins. The small air flow length rather marks them out as a special design case and the subject deserves separate attention. It is not covered in this text.

Segmentally baffled shell-and-tube designs

Segmentally baffled and disc-and-doughnut baffled shell-and-tube designs are not specifically included because the exchanger core may not have sufficiently regular flow geometry. However, there have been some attempts to develop a direct-sizing approach for these exchangers, plus helically baffled shell-and-tube exchangers which are referenced in Chapter 7.

Single-spiral radial flow

Also excluded is the single-spiral heat exchanger with inward and outward spiral (pseudo-radial) flow. Papers analysing performance of this exchanger design have been published by Bes & Roetzel (1991, 1992, 1993). The omission of this design is not a criticism of its usefulness, for in the right application such exchangers may be more economic, or more suitable for corrosive or fouling service.

Extensions

Exchangers that may be suitable for direct-sizing include:

Single-spiral axial design

The single-spiral exchanger with axial flow has been realized and is a candidate for direct-sizing using the thermal design approach outlined in Chapter 4 (Oswald *et al.*, 1999).

Plate–frame designs

The plate-and-frame heat exchanger is not specifically considered, because steady-state design follows standard contraflow or parallel-flow procedures. It is only necessary to source sets of heat-transfer and flow-friction correlations before proceeding.

Plate-and-frame designs can be similar in flow arrangement to plate-fin designs, but there is restriction on the headering geometry. Optimization may proceed in a similar way as for compact plate–fin heat exchangers, but is likely to be less comprehensive until universal correlations for the best plate–panel corrugations become available. The text by Hewitt *et al.* (1994) provides an introduction to steady-state design using plates with standard corrugations, and provides further references. The paper by Focke (1985) considers asymmetrically corrugated plates.

Inlet and return headering for plate-and-frame designs, and the same arrangement for plate–fin designs, may add a phase shift to the outlet transient response following an inlet disturbance. Effects of this headering arrangement have been considered by Das & Roetzel (1995). Faster response is obtained with U-type headering than with Z-type headering, and the choice of U-type headering is evident in the paper by Crisalli & Parker (1993) describing a recuperated gas-turbine plant using plate–fin heat exchangers. However, the reader should consider Dow's (1950) approach to the design of headers in Chapter 8 of this text.

Printed-circuit heat exchangers
These are constructed first by taking a suitable flat plate, then printing a chemically resistant photographic image of material between desired flow channels on to the plate, and then etching the plate to a depth not exceeding 2.0 mm. For the second fluid a further plate with similar etched channels, but probably of different design, is placed on top of the first plate, and the stacking process repeated until a desired stack height is reached. The stack of plates is then diffusion bonded together to form the single core of an exchanger.

Two-stream and multi-stream exchangers may be constructed in this way. It is important that the best geometry of flow channel is selected for each fluid stream, and that proper consideration is given to inlet and outlet headers so as not to create an exchanger with mixed crossflow and contraflow features, as it then becomes problematic to calculate correct temperature profiles.

Depending on geometry and availability of appropriate heat-transfer and flow-friction correlations, thermal design can be approached in the same way as for plate–fin exchangers.

Lamella heat exchangers
Flat tube ducts are fitted inside a tubular shell, leaving equal spacing for shell-side flow between the flat tube ducts. The geometry offers a very flexible surface arrangement, with good means for header connections to shell- and tube-side flow.

Rapid prototyping (but real) designs
The technique of producing rapid prototypes of complex components has now been extended to include construction of complete heat exchangers (see UK Patent GB2338293). The technique involves slicing the finished concept drawings into flat shapes which then may be either cut from meta sheet by laser, or stamped out. These metal sections are then stacked and diffusion bonded to recover the final exchanger. Small ligaments may be required to locate otherwise unsupported parts of a slice in place. If adjacent slices also require support, then ligaments are staggered to preserve flow paths past the ligaments. This approach has already been successful in creating a small and well-designed shell-and-tube heat exchanger, in which baffle passes are repeated to minimize the number of slices required.

Porous metal developments
New interest has been noted in the use of porous, foamed metal fillings inside tubes, and sometimes as external fins. Potential advantages which can be identified include greater metal/fluid surface area for heat transfer, and the possibility of using the porous substrate for mounting catalysts.

1.3 Helical-tube, multi-start coil

This design shown in Fig. 1.1 has no internal baffle leakage problems, it permits uninterrupted crossflow through the tube bank for high heat-transfer coefficients, and provides advantageous counterflow terminal temperature distribution in the

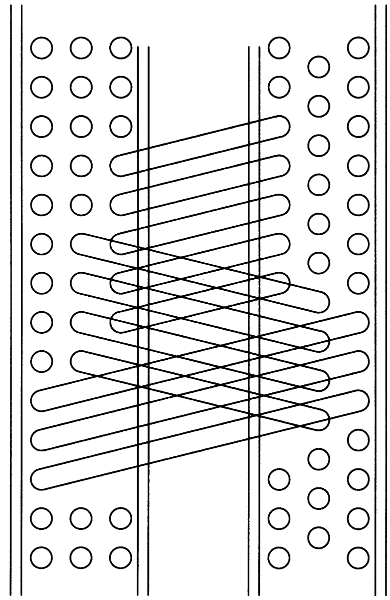

Fig.1.1 Helical-tube multi-start coil exchanger

whole exchanger. Some modification to the log mean temperature difference (LMTD) is necessary when the number of tube turns is less than about ten and this analysis has been provided by Hausen (1950, 1983) in both his German and his English texts.

Although exchangers of this type had been in use since the first patents by Hampson (1895) and L'Air Liquide (1934), consistent geometry in the coiled tube bundle does not seem to have been known before Smith (1960). Since that time programmes of work on helical-coil tube bundles have appeared (Gilli, 1965; Smith & Coombs, 1972; Smith & King, 1978; Gill *et al.*, 1983), and a method of direct-sizing has been obtained by Smith (1986) which is further reported in this text.

Cryogenic heat exchangers to this design have been built by Linde AG and are illustrated in both editions of Hausen (1950, 1983), further examples being found in the papers by Abadzic & Scholz (1972), Bourguet (1972) and Weimer & Hartzog (1972). High-temperature nuclear heat exchangers have been constructed in very large multiple units by Babcock Power Ltd for two AGR reactors (Perrin, 1976), and by Sulzer and others for several HTGR reactors (Kalin, 1969; Profos, 1970; Bachmann, 1975; Chen, 1978; Anon, 1979). A single unit may exceed 18 m in length and 25 tonnes in mass with a rating of 125 MWt.

The pressurized-water reactor (PWR) nuclear ship Otto Hahn was provided with a helical-coil integral boiler built by Deutsche Babcock (Ulken, 1971). For LNG applications, Weimer & Hartzog (1972) report that coiled heat exchangers are preferred for reduced sensitivity to flow maldistribution. Not all of the above heat exchangers have consistent geometry within the tube bundle.

1.4 Plate–fin exchangers

The compact plate–fin exchanger is now well known due to the work of Kays & London (1964), London & Shah (1968), and many others. It is manufactured in several countries, and its principal use has been in cryogenics and in aerospace where high performance with low mass and volume are important. Constructional materials include aluminium alloys, nickel, stainless steel, and titanium. The lay-up is a stack of plates and finned surfaces which are either brazed or diffusion bonded together. Flat plates separate the two fluids, to which the finned surfaces are attached. The finned surfaces are generally made from folded and cut sheet and serve both as spacers separating adjacent plates, and as providers of channels in which the fluids may flow [Fig 1.2(a)].

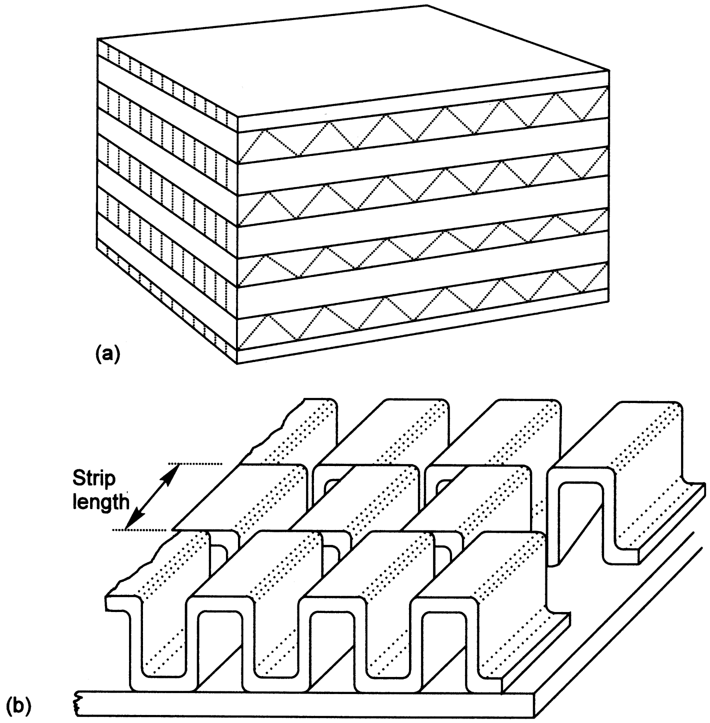

Fig.1.2 (a) Compact plate–fin heat exchanger; (b) rectangular offset strip-fin surface

Many types of finned surface have been tested, see e.g. Kays & London (1964) and Fig. 1.2(b) shows an example of a rectangular offset strip-fin surface which is one of the best-performing geometries. The objective is to obtain high heat-transfer coefficients without correspondingly increased pressure-loss penalties. As the strip-fins act as flat plates in the flowing fluid, each new edge starts a new boundary layer which is very thin, thus high heat-transfer coefficients are obtained.

1.5 RODbaffle

The RODbaffle exchanger is essentially a shell-and-tube exchanger with conventional plate-baffles (segmental or disc-and-doughnut) replaced by grids of rods. Unlike plate-baffles, RODbaffle sections extend over the full transverse cross-section of the exchanger.

Originally the design was produced to eliminate tube failure due to transverse vortex-shedding-induced vibration of unsupported tubes in crossflow (Eilers & Small, 1973), but the new configuration also provided enhanced performance and has been developed further by Gentry (1990) and others.

Square pitching of the tube bundle is considered the most practicable with ROD-baffles, and circular rods are placed between alternate tubes to maintain spacing. To

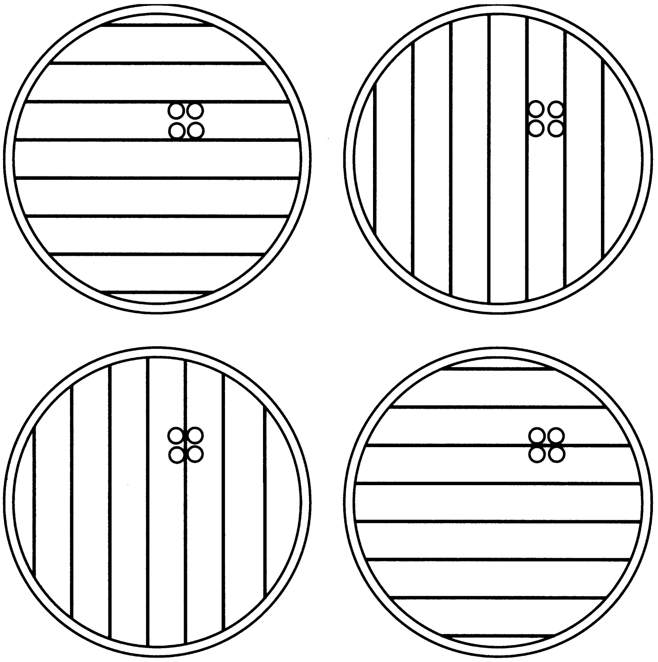

Fig.1.3 RODbaffle set of four baffles

minimize blockage, one set of vertical rods in a baffle section is placed between every second row of tubes. At the next baffle section the vertical rods are placed in the alternate gaps between tubes not previously filled at the first baffle section. The next two baffle sections have horizontal rod spacers, similarly arranged. Thus each tube in the bank receives support along its length.

It might be argued that the RODbaffle geometry is not completely consistent throughout its shell-side, and that it should not therefore be included in this study. However, the spacing rods in the shell-side fluid were found to be shedding von Karman vortex streets *longitudinally* which persist up to the next baffle rod. Thus as far as the shell-side fluid is concerned there is consistent geometry in the exchanger even though the RODbaffles themselves are placed 150 mm apart.

Tube counts are possible for square pitching using the Phadke (1984) approach. Figure 1.3 illustrates arrangement of baffles in the RODbaffle design.

1.6 Helically twisted flattened tube

This compact shell-and-tube design was developed by Dzyubenko *et al.* (1990) for aerospace use, and it complies with the requirement of consistent local geometry in every respect when triangular pitching is used. The outside of the tube bundle requires a shield to ensure correct shell-side flow geometry, and the space between the exchanger pressure shell and the shield can be filled with internal insulating material. The design is illustrated in Fig. 1.4, and the performance of this design is discussed thoroughly in the recent textbook by Dzyubenko *et al.* (1990), although the title of the book is somewhat misleading. Tube counts on triangular pitching are possible using the Phadke (1984) approach.

1.7 Spirally wire-wrapped

A further shell-and-tube concept is based on providing spiral wire-wraps to plain tubes – a concept used with nuclear fuel rods. With triangular pitching it is possible

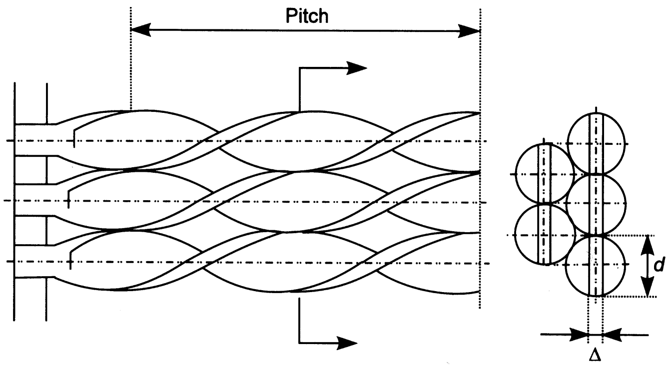

Fig.1.4 Helically twisted flattened tube

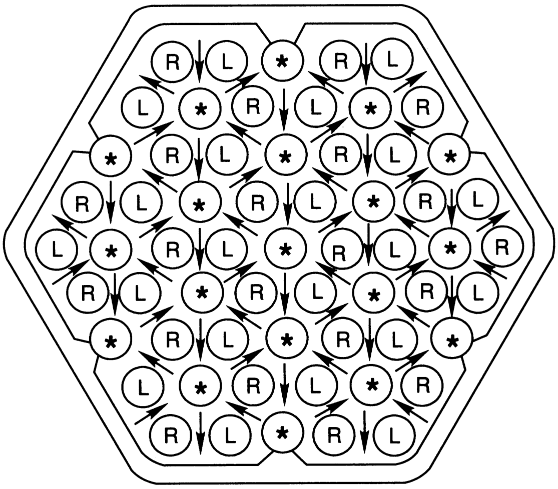

Fig.1.5 Cross-section of R–0–L spirally wire-wrapped layout

to arrange a mixture of right-hand (R), plain (0) and left-hand (L) wire-wraps so as to reinforce mixing in the shell-side fluid. This concept has not been tested for heat exchangers, and it does not quite fulfil the requirements of consistent local geometry, as the plain tubes lack the finning effect of the wire-wrap. The cross-section of a tube bundle is shown in Fig. 1.5, and the wire-wraps extend for about the central 90 per cent of the tube length.

The most common spiral wire-wrap configuration is to have *all* nuclear fuel rods with the same-handed spiral. This leads to opposing streams at the point of closest approach of rods, and swirling in the truncated triangular cusped flow principal flow channels. The spiral wrap is slow, being of the order $12-18^{\circ}$ to the longitudinal axis of the rod. In several nuclear fuel rod geometries the arrangement of rods does not follow a regular triangular pattern, and correlations need to be assessed accordingly.

The R–0–L configuration provides even shell-side fluid distribution and mixing. The Phadke (1984) tube-count method will apply to triangular pitching.

1.8 Bayonet tube

Both bayonet-tube and double-pipe heat exchangers satisfy the concept of consistent shell-side and tube-side geometry, both have been discussed in other works, e.g. Martin (1992). Hurd (1946) appears to be the first to have analysed the performance of the bayonet-tube heat exchanger, but his analysis was not complete and further results are reported in the present text. The upper diagram in Fig. 1.6 show a typical exchanger. Practical uses include heating of batch processing tanks, sometimes with vertical bayonet tubes with condensation of steam in the annuli

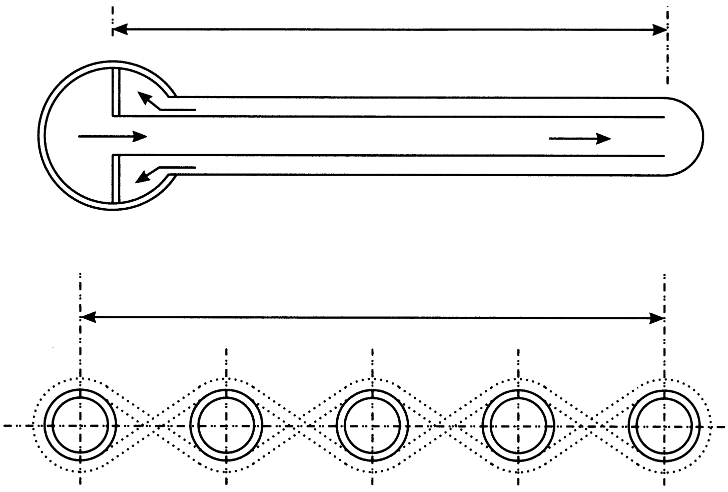

Fig.1.6 Bayonet-tube exchanger (upper diagram). Wire-woven tubes (lower diagram)

(Holger, 1992), freezing of ground, and cooling of cryogenic storage tanks, and high-temperature recuperators using silicon carbide tubes.

Residence time of the fluid in the annulus may be extended by adding a spiral wire-wrap to the outside of the inner tube, thus forcing fluid in the annulus to follow a helical path. When this is combined with insulating the inner tubes, improved external heat transfer will result.

1.9 Wire-woven heat exchangers

The concept of fine tubes woven with wire threads into a flat sheet is a recent proposal by Echigo *et al*. (1992). Given the right layout this arrangement could easily qualify for direct-sizing. The lower diagram in Fig. 1.6 shows the arrangement.

1.10 Porous matrix heat exchangers

The surface of the porous matrix heat exchanger described by Hesselgreaves (1995, 1997) is built up from flattened sections of perforated plate, or flattened expanded mesh metal, stacked so that each section is offset half a pitch from its immediate neighbours (Fig. 1.7). The fluid flows in and out of the plane of the fins in its passage through the exchanger, coupled with diverging and converging flow, thus creating a three-dimensional flow field in the matrix. Individual plate thicknesses are much thinner than with conventional plate–fin geometries, presently ranging from 0.137 to 0.38 mm.

The new geometry offers an increased number of 'flat plate' edges to the flow stream, plus greater cross-sectional area for heat to flow towards the channel

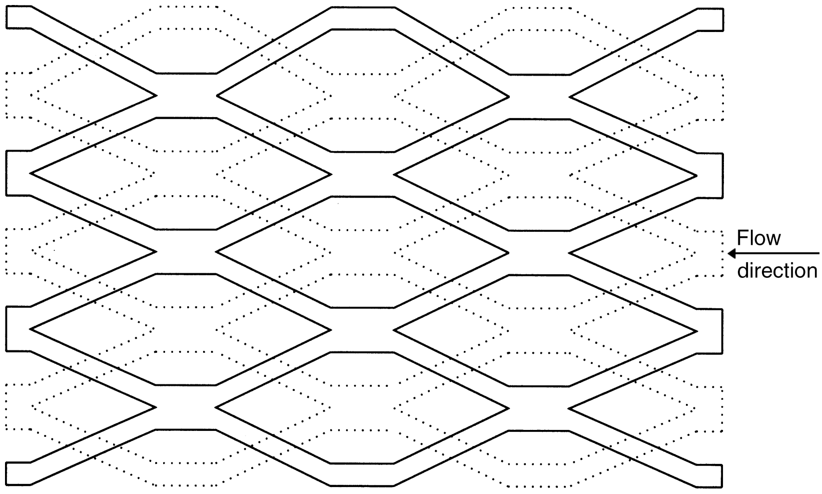

Fig.1.7 Stacked plates of porous matrix heat exchanger

separating plates. The layout is thus better configured for heat transfer than conventional plate–fin geometries. An infinite number of geometries are possible, with the possibility of changing mesh size along the length of the exchanger. Presently only preliminary test results are available, but there is every indication that the pressure loss will be lower, and the heat transfer higher, than for plate–fin designs.

So far, the flattened expanded mesh plates have been diffusion bonded together in packs of from 6 to 15 layers, with separating plates between streams, to form a very strong exchanger. As such a construction seems amenable to forming plate-packs with involute curvature, as illustrated in Fig. 1.8, the prospect of constructing a completely bonded two-pass annular flow exchanger exists. This arrangement could prove suitable for the vehicular gas-turbine application shown in Fig. 1.9.

Sufficient examples of exchangers with a recognizable 'local geometry' have now been given to allow the reader to recognize new types of exchanger which conform to requirements for 'direct-sizing'.

1.11 Some possible applications

At this stage it is only possible to indicate some applications for the heat exchanger configurations described earlier. Not all of this technology is yet in service, or indeed constructed, and the reader is simply asked to appreciate some possible applications which exist for the new direct-sizing designs described.

Propulsion systems

Intercooled and recuperated gas turbine cycles for marine propulsion are presently being developed for the considerable fuel savings that are possible (Cownie, 1993;

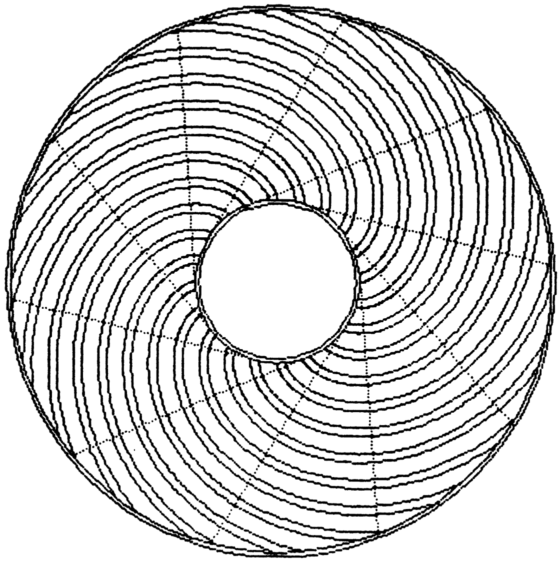

Fig.1.8 Cross-section of involute-curved plate–fin heat exchanger

Crisalli & Parker, 1993). In every case a contraflow heat exchanger arrangement is the natural first choice as it provides more energy recovery than multi-pass cross-flow, but the practicalities of inlet and outlet ducting have also to be considered.

Small recuperators

A recuperator of two-pass involute design has been developed for military tank propulsion (Ward & Holman, 1992), (Fig. 1.8). In the compact vehicle propulsion system shown in Fig. 1.9 (after Collinge, 1994), the power turbine exhaust flows outwards through the exchanger core while high-pressure combustion air flows axially through the exchanger in two passes. Swirling exhaust gases can be directed by an outlet scroll before entering the exhaust stack. Some development work would be required to realize the involute-curved plate–fin exchanger. Thermal sizing is identical to that for the compact flat-plate design. Plate spacing on the high-pressure cold air side is narrow while the spacing on the low-pressure hot gas side is wide.

For a single-pass contraflow design some thought would be required in the arrangement of headers. A further problem with the involute exchanger is the difficulty of cleaning curved channels. Wilson (1995) believes that a rotating ceramic regenerator should be preferred, as it could be more easily cleaned, but it introduces the problem of sliding seals.

Large recuperators

For the recuperator of a larger gas turbine a plate-and-frame design with U-type headering was developed for marine propulsion (Valenti, 1995). Crisalli & Parker

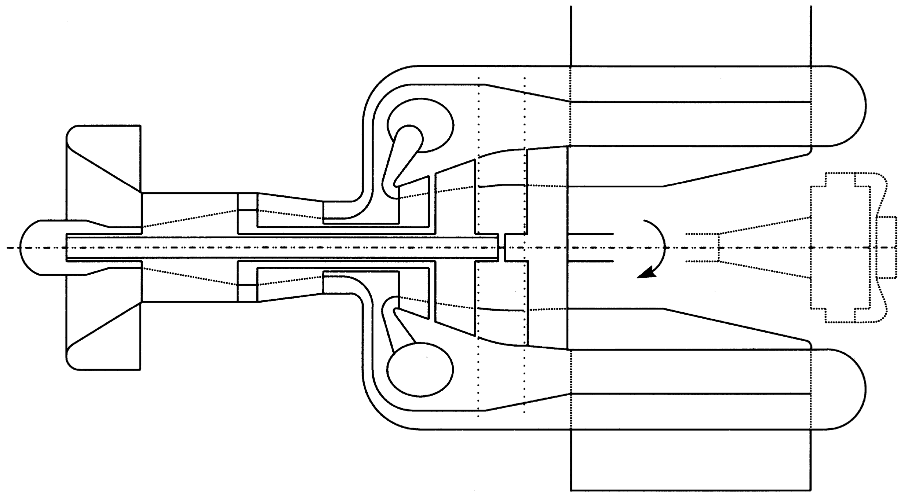

Fig.1.9 Schematic arrangement of two-spool power gas turbine with two-pass cross-flow exhaust exchanger

(1993) and Shepard *et al.* (1995) report on development of the WR-21 intercooled and recuperated marine gas turbine. In the latest Rolls-Royce design of recuperator, compact plate–fin-type surfaces were used.

Intercoolers

An intercooler can also be fitted between low-pressure and high-pressure compressors of large marine and land-based systems (Crisalli & Parker, 1993; Bannister *et al.*, 1994). This has to be a contraflow plate–fin design for compactness, with fresh-water/glycol supply and return from external annular header pipes. The exchanger can be segmented for ease of maintenance. With this arrangement the gas turbine would not become exposed to sea-water leaking from a damaged intercooler. Pressure in the closed-loop fresh-water/glycol system can be adjusted to suit the desired operating conditions.

Liquid hydrogen propulsion

Related to both the helical-tube, multi-start coil heat exchanger and the involute-curved plate–fin exchanger is the involute-curved, serpentine-tube panel design used by Pratt & Whitney in one of their experimental engines powered by liquid hydrogen (Mulready, 1979). The tube panel is a single serpentine tube arranged such that the shell-side fluid may flow transversely over the tube as in crossflow over rows of tubes, Fig. 1.10(a).

Each panel is given an involute curve and is placed together with others in an annular pattern as in Fig. 1.10(b), the pitch between adjacent panels is constant with radius and the shell-side fluid sees the same geometry everywhere. Thermal

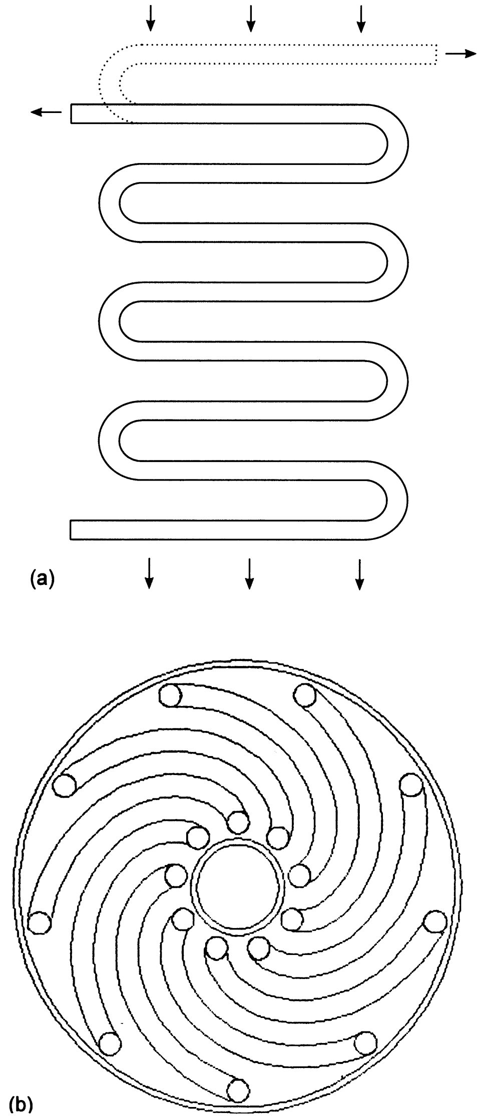

Fig.1.10 (a) Serpentine tube panel; (b) involute pattern

performance of flat serpentine panels has been discussed by Hausen (1950), and will apply to involute-curved serpentine panels also. The only difference is a secondary effect due to involute curvature, which may affect tube-side heat transfer and pressure loss hardly at all. The incentive to go to this design must be high, to make pressure loss in repeated bends acceptable.

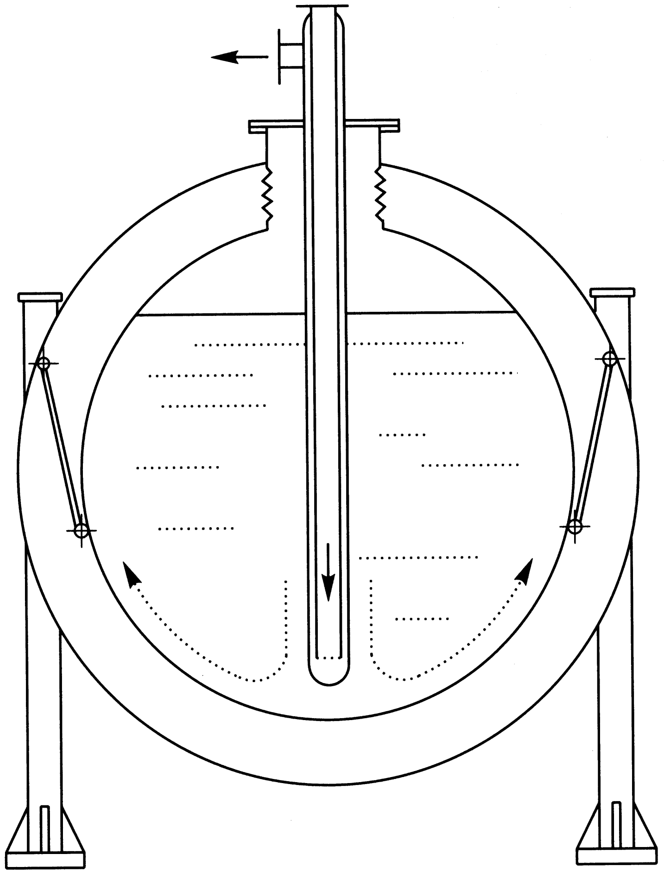

Fig.1.11 Cryogenic storage tank with bayonet-tube exchanger (schematic)

Cryogenic plant

Cryogenic heat exchangers

The sizing of cryogenic heat exchangers is discussed in Chapter 11. A step-wise rating approach is required, especially when near to the critical point, and once that technique has been developed it is straightforward to apply the same methods to cryogenic designs in which the LMTD concept would otherwise be perfectly viable.

Cryogenic storage tank

One problem that has troubled cryogenic and petrochemical industries is that of 'roll-over' in cryogenic storage tanks. The liquid cryogen is at a higher pressure at the bottom of the tank compared with the surface, due to liquid density. The saturation pressure is thus higher at the bottom of the tank. If there is external heat leak into the tank then it is possible for the liquid at the bottom of the tank to be at a temperature higher than that at the top. If conditions are such that this liquid travels to the top of the tank by convection then the massive evaporation which ensues at the lower pressure can be sufficient to rupture the tank.

Presently a mixing propeller on a shaft is used to circulate liquid within the tank to keep the contents close to isothermal.

The bayonet-tube heat exchanger is a design that requires only single penetration of a pressure vessel. If such an exchanger is fitted to the top of a cryogenic storage tank and a cryogen used in this exchanger to cool the contents of the tank, then controlled circulation may be set up in the tank with colder fluid at the centre falling to the bottom and warmer fluid at the side walls rising to the free surface (Fig. 1.11.) An extension of this would be to provide the mixing propeller with a hollow shaft incorporating a bayonet-tube exchanger. The possible effectiveness of the bayonet-tube exchanger in inhibiting 'roll-over' seems worthy of investigation.

It is hoped that the above long-range concepts may stimulate the reader to consider other arrangements for heat exchangers that can be directly sized.

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